VENTILATION OF DAIRY BARNS

by

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INTRODUCTION

Ventilation of animal shelters presents a problem, mainly because insufficient heat is produced by the animals to maintain a comfortable temperature within the shelter. If the air movement is sufficient to remove moisture given off by the animals then the temperature is too low. If the air movement is restricted so as to maintain adequate temperatures, the amount of moisture removal is too little and condensation occurs, causing an unsanitary condition and rotting and rusting of the structure.

Insulation is provided to increase the amount of heat available for ventilation purposes. This is quite effective, and if the structure is properly designed may be entirely adequate, but some structures which have been built without proper consideration of the ventilation problem are difficult to insulate properly. The Kansas State dairy barn is a structure which is in this category.

Dairy barn ventilation is becoming of less importance because many dairymen, except in the extremely cold climates, are using the open shed type of housing. This eliminates the ventilation problem where this is done, but many dairymen who already have a good stanchion barn are hesitant to change to open shed housing. Also in the colder climates even if loose housing is used it is done in a closed barn. There are also buildings such as the Kansas State dairy barn where feeding and other experiments are conducted on the animals, and it is

essential that the cows be housed. In all of these cases there is a ventilation problem.

General Considerations

In dairy barn ventilation at least three general considerations should be made:

1. Health, comfort and maximum production of the animals with a minimum of feed consumption.

2. Factors effecting the length of life and usefulness of the structure.

3. The comfort and efficiency of the workmen in the building.

Health, Comfort and Production of Cows. Environmental conditions which will give the maximum desired in health, comfort and production of cows are not exactly known, but authorities (8,12,16,17) are generally agreed that near 50° F. is the optimum temperature for maximum production of dairy cows. Ragsdale et al. (12) state that the optimal temperature zone for quantity and efficiency of milk production appears to be not too far from 50° F. Lowering of temperatures from 50° to 4° F. increased feed consumption and butterfat percentage, but decreased somewhat the milk yield. There seems to be a greater decrease with temperatures above 50° F. than there is at temperatures below 50° F., thus it is probably more important to keep temperatures below 50° F. than it is to attempt to maintain them as high as 50° F.

Length of Life and Usefulness of the Structure. Moisture conditions have more effect on the structure than other environmental conditions. Condensation of moisture on the walls and ceiling will tend to shorten the length of life of a structure by causing the wood to rot and the metal to rust. Moisture absorption by masonry will cause it to deteriorate more rapidly, especially when freezing and thawing occurs. Moisture on the walls and ceiling is also an unsanitary condition as it encourages growth of fungi and accumulation of dust and dirt where it is wet. Temperatures high enough to prevent freezing of water fixtures add to the usefulness of the building. When temperatures are below the freezing point either water facilities cannot be used in the barn or special precautions have to be taken to prevent their freezing.

Comfort and Efficiency of the Workmen. Feeding, milking and other chores are more efficiently done when temperatures and humidity are such that the workmen are comfortable. Either cold temperatures or damp conditions cause the workmen to slight some of the chores in an effort to get through quicker so they will be able to get into a more comfortable environment. When they are comfortable they can do their chores more efficiently as they will spend less time providing for their own comfort.

Purpose of Investigation

This investigation was conducted for the purpose of determining whether a heat exchanger could be adapted for use in ventilation of the Kansas State dairy barn. The proposal was to have a heat exchanger designed so as to utilize heat from the exhaust air to warm the incoming fresh air to make it possible for a larger quantity of air to be circulated without reducing the temperature of the barn below that desired. This would remove more of the moisture from the barn and reduce or prevent some of the condensation. If a practical heat exchanger could be put into use much of the heat now lost in the exhaust air could perhaps be utilized in the barn to maintain higher temperatures.

The 70 cows in the barn give off approximately 200,000 Btu per hour of sensible heat. This amount of heat, if it had to be provided by burning fuel would require considerable expense. Obviously all of the heat given off by the cows cannot be saved, but if a small amount of it could be made available, additional ventilation could be provided and still make it possible to maintain desired temperatures in the barn. The feasibility of saving the heat by this method would depend on whether the additional cost involved in providing equipment necessary to save the heat would be less than the cost of providing it by some other method.

The investigation was divided into two parts; first, a

study of the actual conditions as they are in the Kansas State dairy barn and temperature differences between the inside and outside of the barn, and second, a study of a plate heat exchanger to determine the amount of heat which can be transfered from the exhaust air to the incoming fresh air.

The investigations were conducted during the winter of 1949-50. This was a very mild winter and because there were no long cold or stormy periods, the information obtained as to conditions in the dairy barn was rather incomplete. However, as the data obtained agree very closely with the calculated data, they were considered sufficient to use in the study.

REVIEW OF LITERATURE

A great deal of work has been done on the subject of ventilation of dairy stables, and also on the use of heat exchangers. However, no data on work done to use a heat exchanger in dairy barn ventilation were found, but several had suggested the possibility of doing this. A large portion of the work done on heat exchangers was at much higher temperatures than encountered in dairy barns.

Ventilation of Dairy Barns

The pioneer in farm building ventilation was F. H. King (9). He designed the famous King system of gravity ventilation which used the heat produced by animals to furnish

motive power for movement of air. His design was based on a heat output of 76,133 Btu per cow per day. Through his experiments he demonstrated that lack of ventilation had harmful effects on the general health of dairy cows. He set a standard of air purity which permitted not more than 3 per cent of breathed air in the stable at any time. This standard required a movement of some 3600 cubic feet per hour per cow which resulted in low barn temperatures if the barn was not well insulated. He believed that moisture produced by the cows should be removed from the stable which will be accomplished if the requirement regarding breathed air is met. Figures of heat and moisture production of animals were not available until Armsby and Kriss (1) published the results of their work on this subject in 1921. They calculated that heat given off varied from 65,000 Btu per day for a Jersey producing 20 pounds of milk to 88,000 Btu per day for a Holstein cow producing 45 pounds of milk. This includes the latent heat of the vaporized water which cannot be regarded as available for ventilation purposes unless some condensation takes place. They also found that this latent heat of water vapor constituted approximately 25 per cent of the total heat given off by the animals. They subjected cattle to far greater concentrations of carbon dioxide than the limit of 0.167 per cent volume set by King (9) without serious effect on the cattle. They considered their results to be only approximate and suggested that

further work be done on this line.

Others (1, 15) felt that the use of figures obtained by Armsby and Kriss (1) would lead to error if used because heat and moisture production vary with temperatures. The amount of heat given off varies between 2,000 and 3,500 Btu per cow depending on the size of the cow and the ambient temperature and humidity conditions existing at the time the tests were made (17).

Kelley and Rupel (8) concluded that optimum stable temperatures for cows in stanchions under Wisconsin winter conditions to be about 50° F. and within the limits of 45° and 65° F., changes of 10° F. in the controlled stable affected principally the first three milkings. Suggestions made by Kelley (7) are that temperatures between 40° and 60° F. are desirable for dairy cows. He also showed that having higher temperatures in the barn required a smaller amount of ventilation for moisture removal. Teasdale (16) gave the optimum temperature in dairy barns as being between 45° and 50° F. In an Editorial in Agricultural Engineering of May, 1939, there was cited a need for research to determine optimum comfort conditions for cows with reference to temperature, humidity, and air movements.

The amount of moisture to be removed by ventilation is from 11-18 pounds per day (14) per cow. Teasdale (16) gives the desired relative humidity as 60 per cent, while Fairbanks (5) lists 75 to 80 per cent as desirable. Many writers do

not list a definite percentage for relative humidity, but they are generally agreed that it should be sufficiently low to prevent condensation on the walls most of the time.

The air movement is given by Mitchell (11), Fairbanks (5) and Carter and Foster (2) as being 60 cfm per cow. This is sufficient to keep the dilution of air below the limits as set by King (9). However, calculations are necessary to determine the amount of air movement necessary to remove moisture from the barn, these being made at the design conditions for the particular barn under consideration. Kelley and Rupel (8) concluded that there was a direct relationship between excessive drafts and pneumonia. Risk of sickness appears to increase with the degree of exposure to drafts. They also concluded that artificial heat in stables was undesirable.

Clyde (3) suggests that a suitable means should be provided to save some of the large amount of heat lost in ventilation of dairy stables. He stated that some method such as having concentric pipes with either cold or warm air in the small pipe and the other flowing in the annular space, or that a heat exchange unit be incorporated in the walls or ceiling of the structure. The saving of heat in this manner would permit higher temperatures in the barn or allow for additional ventilation with subsequent reduction of moisture.

Heat Exchanger

Much work has been done on the exchange of heat from a warm fluid to a colder one. Generally the temperature range and temperature differences are much greater than encountered in dairy barn ventilation, and the fluid used is more often liquid than gases. Another difference in the exchanger used in this study and the ones usually found in practice is that the one used here has flat plates separating the warm air from the cold, and in most exchangers one of the fluids flows in pipes or tubes.

Reynolds (13) in 1874 suggested the use of mass velocity to determine the heat transfer coefficient. McAdams (10) shows a plot of Nusselt's data for three different gases flowing in a tube having an inside diameter of 0.868 inch where h is plotted versus G on logarithmic coordinates, except in the very low range where natural convection increases h, the plotted points are very close to the curve and the slope is equal to 0.8 or h varied as $G^{0.8}$. It was found that G had the same effect regardless of whether in the product $G = V\rho$, average linear velocity V was high and density ρ low, or the reverse occured.

Considering the factors in the basic energy and hydromechanical equations wherein

$$h = \alpha G^a D^b c^d \mu^e k^f \qquad (1)$$

Table 1. Nomenclature.

a, b, d, e, and f	Dimensionless constants.
c	Specific heat, Btu/(lb of fluid)(deg F).
D	Diameter of conduit, feet.
G	Mass Velocity, lb of fluid/(sec)(sq ft of cross section), equals V .
H	Total heat transfer, Btu.
h	Coefficient of heat transfer between fluid and surface, Btu/(hr)(sq ft)(deg F).
i	Enthalpy, Btu/lb of fluid.
K	Dimensionless constant.
k	Thermal conductivity of fluid, Btu/(hr)(sq ft)(deg F per ft).
L	Length, feet.
N_n	Nusselt number, dimensionless, equals hD/k .
R_n	Reynolds number, dimensionless, equals DG/μ, equals $VD\rho/\mu$.
U	Over-all coefficient of heat transfer, Btu/(hr)(sq ft)(deg F).
V	Velocity, feet/second.
α	Proportionality factor, dimensionless.
ρ	Density, pounds fluid per cubic foot.
μ	Absolute viscosity of fluid, lb/(sec)(ft).

a dimensional analysis gives

$$\frac{hD}{k} = \alpha\left(\frac{DG}{\mu}\right)^{a}\left(\frac{c\mu}{k}\right)^{b} \tag{2}$$

McAdams (10) gives the equation for heating and cooling of fluids inside tubes as

$$\frac{hD}{k} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k}\right)^{0.4} \tag{3}$$

Inasmuch as $c\mu/k$ varies but little for any given fluid he gives a simplified equation for equation (3)

$$\frac{hD}{k} = 0.020\left(\frac{DG}{\mu}\right)^{0.8} \tag{4}$$

McAdams (10) also gives another simplified equation for the same condition

$$h = 0.0144\left(\frac{cG}{D^{0.2}}\right) \tag{5}$$

Colburn (4) suggests plotting $\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{2/3}$ versus $\left(\frac{DG}{\mu}\right)$ on logarithmic coordinates. He also gives the equation

$$\left(\frac{h}{cG}\right)\left(\frac{c\mu}{k}\right)^{2/3} = 0.036\left(\frac{LG}{\mu}\right)^{-0.2} \tag{6}$$

for fluids flowing parallel to flat plates. McAdams (10) gives a similiar equation without giving the values of the

constant or the exponent. The difference in this equation and the one shown previously is the characteristic dimension in the Reynolds number where L (length) is used instead of D (diameter).

McAdams (10) reports that if condensation occurs the film coefficient will be higher than if there is no condensation.

TEST PROCEDURE

Tests were made on a model heat exchanger to determine the coefficient of heat transfer when operating at conditions which may be encountered in dairy barn ventilation, and temperature readings were taken in the dairy barn to find the amount of heat lost from the barn.

The model heat exchanger built was of the flat plate type, with air moving in counterflow through rectangular ducts. The arrangement of the exchanger was such that exhaust air and fresh air flowed in alternate ducts. The exchanger used had a total of 17 ducts with exhaust air flowing in 9 of them and fresh air flowing in the other 8 ducts. Air was pushed through the exchanger with a fan on each end, with the fan blowing exhaust air located inside the room and the one blowing fresh air from outside the building. The exchanger ducts were made of flat aluminum sheets with wooden strips 1 x 3/4 inch forming the sides of the ducts and separating the aluminum sheets. The interior dimensions of

each duct are approximately 44 x 15 x 3/4 inches. This makes a total area of heat transfer surface equal to 80 square feet.

Aluminum was used, not because of any apparent advantage over sheet steel or other metals, but because it was the only metal available at the time the exchanger was built. The outside of the exchanger was covered with plywood. Air was brought into the exchanger from the fans through canvas sleeves. At the exit end of both the exhaust air and the fresh air a 6 x 6 inch duct was attached for the air to flow through so that velocity measurements could be made. Sliding doors and windows were made on the exchanger for the purpose of making observations and inserting instruments for measurements.

In making a test on the exchanger, outside air was blown through one side and room air through the other. Air velocity measurements were made with a pitot-static tube with measurements taken at the centers of nine equal squares in the 6 x 6 inch duct. An average of the nine measurements was used to calculate the velocity pressure. The pressure difference of the pitot-static tube was measured with an inclined draft gauge. Using the average of the pressures measured the air velocity was obtained from the basic equation

$$V = \sqrt{2gh} \tag{7}$$

where V is feet per second and h is feet head of the flowing fluid. In feet per minute this equation becomes

$$V_m = 1096.5 \sqrt{\frac{h_v}{d_a}} \qquad (8)$$

where h_v is the velocity pressure from the indicating manometer in inches of water and d_a is the density of the air flowing in the duct in pounds per cubic feet. The density of the air was calculated by using the equation

$$d_a = 1.325 \frac{P_b}{T} \qquad (9)$$

where P_b is the barometric pressure in inches of mercury and T is the absolute temperature in degrees Fahrenheit. From the velocity, density and cross sectional area of the duct the pounds per hour of air going through the exchanger were calculated.

Wet and dry bulb temperature readings were taken with a bulb type aspirator psychrometer, of both the exhaust and fresh as they entered and left the exchanger. Knowing the mass of air and the wet and dry bulb temperatures, the heat content of the air as it entered and left the exchanger was calculated by the equation

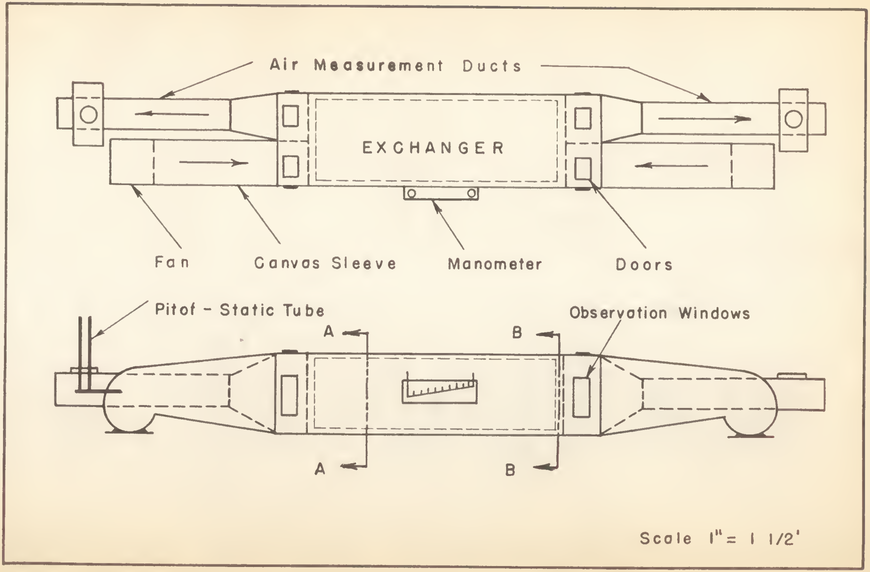
$$h = c_p t_d + W_s \left(1059.2 + 0.45 t_d \right) \qquad (10)$$

The heat loss from the exhaust air should equal the heat gain by the fresh air.

EXPLANATION OF PLATE I

This schematic diagram of the experimental heat exchanger shows the location of air measurement ducts, fans and apparatus used in making measurements. Arrows on ducts indicate direction of air flow.

PLATE I

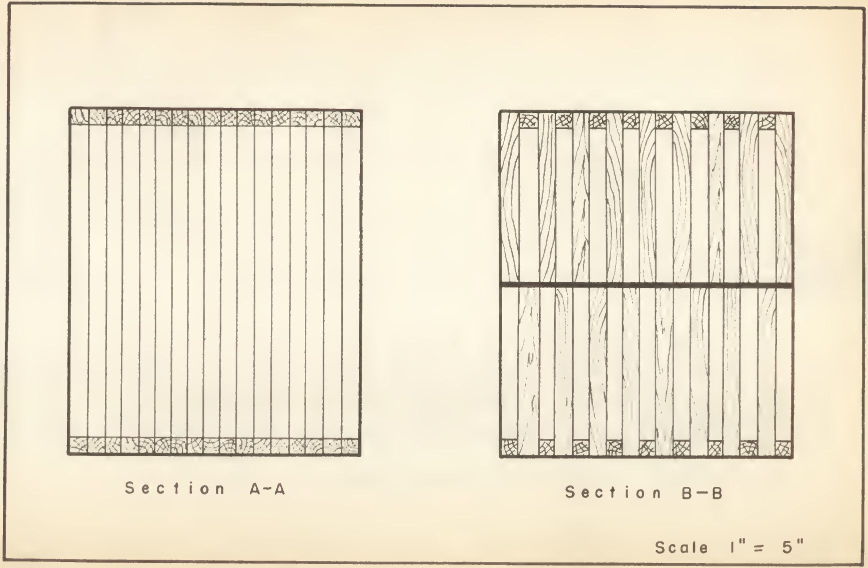

Air Measurement Ducts
EXCHANGER
Fan
Canvas Sleeve
Manometer
Doors
Pitof - Static Tube
Observation Windows
A
A
B
B
Scale 1" = 1 1/2'

EXPLANATION OF PLATE II

Section A-A is a central section of the Heat Exchanger, the lines representing the aluminum plates which form the air ducts. Warm air flows in the outside duct, and cold and warm air flows in alternate ducts.

Section B-B is an end section showing how alternate ducts are blocked to keep the warm and cold air separated. Location of both sections is shown in PLATE I.

PLATE II



The coefficient of heat transfer U from one air stream to the other is calculated from the equation

$$Q = UA\,\Delta t_m \tag{11}$$

where Q is the average of the heat lost by the warm air and the heat gained by the cold air.

Inasmuch as several investigators have shown that the coefficient of heat transfer varies with mass velocity, the objective in the tests was to vary the mass of air flowing through the exchanger in order to be able to determine how the coefficient h varies with the mass velocity G. The air velocity was regulated to get a different mass flow in each test, the range of mass velocities being that which might be practical in dairy barn ventilation.

RESULTS OF HEAT EXCHANGER TESTS

Table 2 shows the data obtained from tests made with the heat exchanger. These data were used to determine an equation for this particular exchanger giving the relationship between the film coefficient h, and the mass flow of air G. It was assumed that the equation would take the form of equation (2), and inasmuch as $\left(\frac{c\mu}{k}\right)^d$ is a constant for a given fluid within a limited temperature range, the value was included in the constant K', making the form of the equation

$$\frac{hD}{k} = K'\left(\frac{DG}{\mu}\right)^a \tag{12}$$

Table 2. Heat exchanger test data.

Test no.		Duct vel. fpm	cfm	Lbs. per hour	i_1	i_2	i net	H	U	h	Exch vel. ft/sec	d	N_n	R_n
1	Ex.	1040	260	1110	22.46	20.43	2.03	2250	1.53	2.94	5.88	.071	33.5	6050
	In.	1030	257	1125	16.10	17.90	1.80	2030		3.19	6.53	.073	37.6	6850
2	Ex.	1050	263	1130	21.22	19.56	1.66	1875	1.40	2.73	5.86	.072	31.0	6160
	In.	995	249	1120	15.32	17.03	1.71	1915		2.98	6.32	.075	35.2	6900
3	Ex.	1030	257	1160	15.60	13.54	2.06	2390	1.39	2.74	5.80	.075	31.2	6330
	In.	910	228	1075	8.06	10.22	2.16	2320		2.83	5.80	.078	33.4	6600
4	Ex.	720	180	820	15.17	13.00	2.17	1780	1.05	2.01	4.06	.076	22.7	4480
	In.	690	172	820	7.71	10.00	2.29	1860		2.21	4.37	.079	26.0	5040
5	Ex.	950	237	1050	18.54	16.61	1.93	2020	1.51	2.88	5.36	.074	32.8	5720
	In.	930	232	1060	12.74	14.55	1.81	1920		3.18	5.90	.076	37.4	6500
6	Ex.	945	237	1045	19.12	17.41	1.71	1790	1.50	2.86	5.35	.073	32.6	5700
	In.	920	230	1045	13.39	14.95	1.56	1630		3.16	5.85	.075	37.2	6400
7	Ex.	1060	265	1160	19.64	17.70	1.94	2250	1.34	2.66	6.00	.073	30.2	6330
	In.	920	230	1050	12.78	14.71	1.93	2030		2.70	5.86	.076	31.8	6450
8	Ex.	1005	251	1078	23.35	22.36	0.99	1067	1.64	3.12	5.66	.072	35.6	5880
	In.	995	249	1090	19.12	20.07	0.95	1037		3.46	6.31	.073	40.8	6720
9	Ex.	890	222	980	17.27	15.59	1.68	1645	1.31	2.47	5.02	.074	28.1	5340
	In.	900	225	1020	11.51	13.19	1.68	1710		2.80	5.70	.076	32.9	6260
10	Ex.	1015	254	1120	17.51	16.07	1.44	1610	1.54	3.09	5.73	.073	35.2	6120
	In.	900	225	995	12.36	14.06	1.70	1690		3.07	5.72	.075	36.2	6100
11	Ex.	780	195	860	18.41	16.63	1.78	1530	1.87	3.52	7.93	.074	40.0	8480
	In.	705	176	808	11.39	13.06	1.67	1350		4.00	8.95	.076	47.2	9900

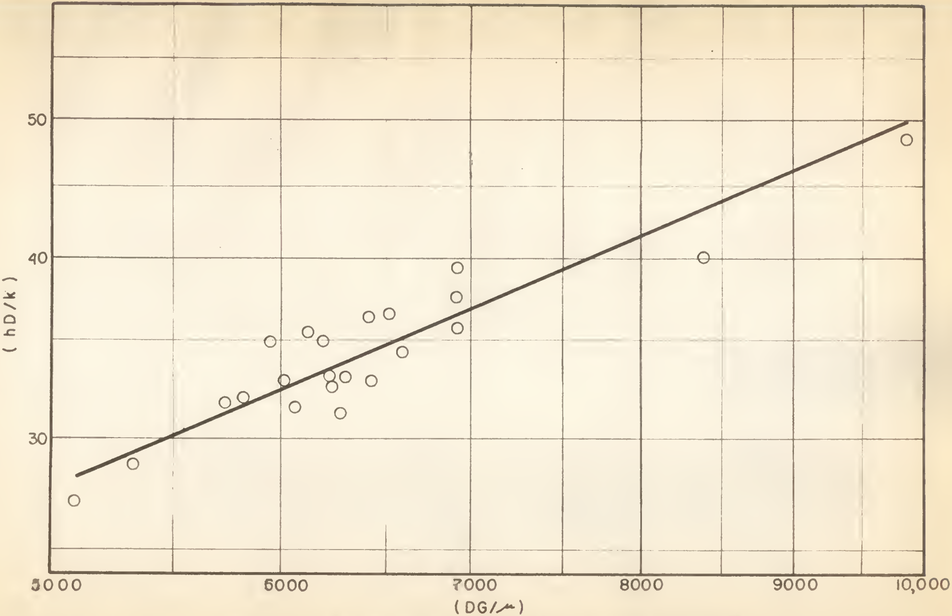


Fig. 1. Plot of experimental data for heat exchanger.

The value U was measured in making the tests. An effort was made in most cases to make the Reynolds number for exhaust air and fresh air the same. This was not accomplished too well and in order to obtain the values of h_1 and h_2 in the equation

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2} \tag{13}$$

the relationship given by McAdams (10) was used, that is h varies a $G^{0.8}$. From this relationship a value of h_1 in terms of h_2 was obtained by using equation (13) as the following sample calculation shows.

Using the values of U and Reynolds numbers as obtained in test No. 4 the values of h_1 and h_2 were obtained in the following manner;

$$h_2 = h_1 \left(\frac{R_{e1}}{R_{e2}}\right)^{0.8} = h_1 \left(\frac{6600}{6330}\right)^{0.8} = 1.032\, h_1$$

then $\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}$ and $\frac{1}{1.39} = \frac{1}{h_1} + \frac{1}{1.032\, h_1}$

$$\frac{1}{1.39} = \left(\frac{1}{1.032} + 1\right)\frac{1}{h_1}$$

$h_1 = 2.74$ and $h_2 = 2.83$

Knowing the value of h, D, and k, Nusselt's number was obtained and this plotted on logarithmic paper versus the Reynolds number. Logarithms of Reynolds and Nusselt's numbers were used, and by the method of least squares an equation

for this exchanger was obtained, the equation being;

$$\frac{hD}{k} = 0.02\left(\frac{DG}{\mu}\right)^{0.85} \tag{14}$$

The data from which this equation were obtained is plotted in Fig. 1.

The correlation coefficient for the logarithms of the numbers about the straight line was 0.856. The line of this equation has the same Y intercept as equation (4) given by Reynolds (10), but the slope is greater, being 0.85 as compared to 0.80. As the relationship that h varies as $G^{0.8}$ was used to obtain the points used in calculating Nusselt's number it was felt that another assumption should be made and recalculate the numbers again, but on recalculation of a few numbers it could be seen that the value of the equation would not be affected. The change in h would not be before the second digit to the right of the decimal.

No attempt was made in the study to determine a coefficient of heat transfer when condensation occurs in the exchanger. According to McAdams (10) when condensation occurs the coefficient of heat transfer increases. There is very apt to be condensation within the exchanger, so it is quite probable that the heat transfer per square foot degree difference will be better than that shown by the equation.

With practical velocities in the heat exchanger, the value of the overall transfer coefficient U from the exhaust

air to the fresh air will vary between 1.0 and 3.0 Btu per hour per square foot per degree Fahrenheit.

DAIRY BARN STUDIES

The data obtained in the dairy barn were for the purpose of determining the temperature difference between the inside and outside which can be maintained in the barn with only the heat produced by the dairy cows.

Conditions in Kansas State Dairy Barn

The barn is constructed mainly of concrete, stone and clay products. All of these have a relatively high coefficient of thermal conductivity and there is no insulation in the walls or ceiling. With this type of structure the heat losses are very high which results in low temperatures inside the building. The rapid flow of heat through the walls and ceiling causes the inside surfaces of the barn to have temperatures considerably below the temperature of the air in the building. Because of this large temperature difference the temperature of the inside surfaces is quite often below the dewpoint temperature of the barn air. Whenever this occurs there will be condensation of moisture on the inside surfaces. There is moisture condensation in the Kansas State dairy barn almost every night during the housing season. During some periods the water drips from the ceiling and also runs down the walls, making it very undesirable inside the

barn. One of the workmen stated that it was necessary to wear his raincoat in the barn most of the winter, even though statements of this kind are not usually accurate, it does indicate that the conditions are very undesirable.

The conditions in this barn are about as favorable for moisture condensation as they possibly could be. It would be possible to circulate sufficient air through the barn to remove moisture and prevent condensation most of the time, but if this were done the barn temperatures would be too low during cold periods for comfort of the cows and the workmen. It would be desirable to eliminate moisture condensation on the walls and ceiling, however, a ventilation system which would accomplish this would be impractical if not impossible unless heat other than from the cows were made available. Teasdale (16) states that complete elimination of condensation on single glass windows is impractical. The same may be true of complete elimination of condensation on the walls and ceiling of the Kansas State dairy barn, because of the conditions which are so favorable for condensation. However, a reduction of the amount of condensation would be very desirable and also possible.

Calculations show that the heat losses from the dairy barn are those listed in Table 3. The weighted average for heat loss from the barn is equal to 0.46 Btu per hour, square foot, degree Fahrenheit. This is slightly higher than the value of 0.43 found by taking temperate readings in the barn

Table 3. Heat loss from barn.

Area	U	Area sq. ft.		Btu per ^{o}F temp. difference per hour
		Heat loss through exposed area		
Ceiling	0.420	6040		2540
Outside walls	0.306	2128		650
Inside walls	0.252	590		150
Windows	1.130	422		475
Doors	0.340	280		95
Total				3910
	Feet of crack	Cu. ft. infiltration per foot of crack	Total cu. ft. infiltration	
		Heat loss through infiltration		
Windows	646	84	54,200	1020
Doors	233	84	19,500	370
Total				1390

and assuming that each cow gives off 2800 Btu heat per hour. Both values are given on the basis of heat lost per degree difference in temperature, but the calculated value does not take into account the fact that the temperature differences between the cow barn and the hay mow, and the cow barn and the rooms on the North and South ends of the barn are much less than the temperature difference between the inside of the cow barn and the outside.

Preliminary temperature readings were made at many random points throughout the barn in order to determine a logical place to install thermometers. It was found from these random readings that the temperature varied little throughout the barn at the same elevations. The variation was in a vertical direction from floor to ceiling. The average temperature in a vertical line was between five and six feet from the floor, consequently the thermometers were placed at that elevation at several locations throughout the barn. Temperature readings were taken at these thermometers at various times. From these readings the average inside temperature was determined and from that and the corresponding outside temperature, the heat losses from the barn were determined by assuming a constant sensible heat output per cow of 2800 Btu per hour. This assumption of constant heat output for the cows is not exactly correct as has been shown by several investigators (1, 15). Work being done at the psychro-energetic laboratory at the University of Missouri

at the present time indicates that this is true, but figures as to how the heat output varies are not available. As 2800 Btu per hour is the value most commonly used, that value was also used in this study.

The data were rather incomplete because of the mild weather which prevailed during the winter which the studies were made, but as the limited data obtained aggreed very closely to the theoretical value it was felt that they were sufficient to use in the study.

Based on the assumption that each cow produced 2800 Btu per hour of sensible heat, a curve of heat losses from the barn in Btu per hour degree difference between outside and inside temperatures was obtained by the method of least squares, this curve is shown in Fig. 2. The overall average coefficient of heat transfer from the barn was found to be 0.43 Btu per hour square foot degree Fahrenheit. The fact that the heat losses per degree difference increases as the outside temperature increases is probably due to the fact that the workmen are not as careful in keeping doors closed as they go through them when the outside temperature is high as they are when it is low. The large amount of heat storage in the walls of the barn has considerable effect on the temperatures in the barn over short periods of changing temperatures. The barn will not cool off nearly as fast as would be the case for a barn having walls in which the heat storage is not as great as the one studied. The reverse would also

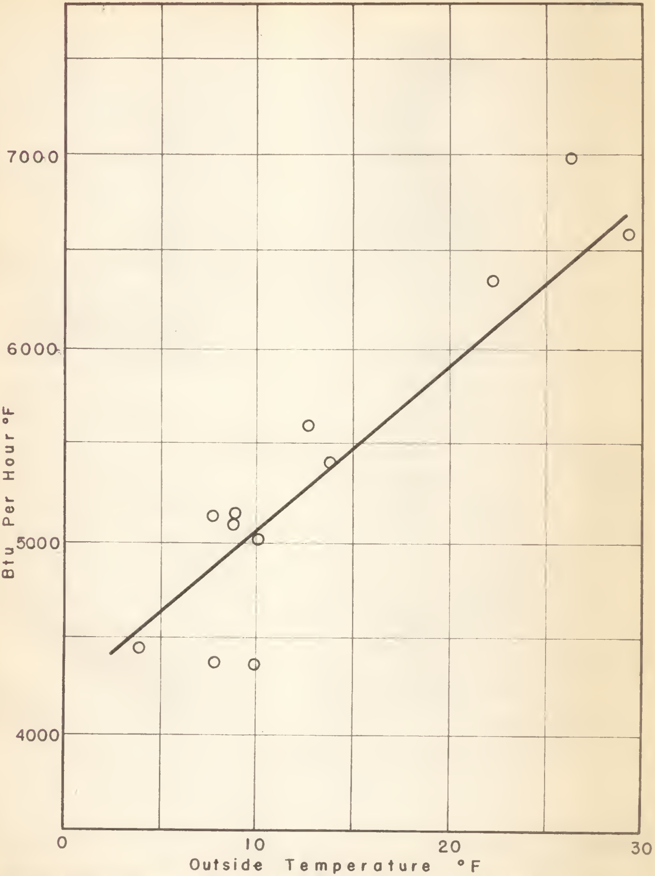


Fig. 2. Btu loss per hour from Kansas State dairy barn at various outside temperatures.

be true as when the outside temperature is rising the walls will absorb heat and the inside temperatures will rise much slower than the outside temperatures. This heat storage is beneficial in smoothing out the peaks of the temperature curves inside the barn, but it has a detrimental effect in increasing the amount of condensation when the temperature is rising. No attempt was made to determine the amount of heat storage by the barn walls.

USE OF HEAT EXCHANGER IN BARN VENTILATION

The purpose of using the heat exchanger in ventilation of the dairy barn is to save some of the heat lost in the exhaust air to warm the incoming cold air. This has two effects on ventilation; first, the resulting inside temperatures will be raised giving greater comfort for the cows and workmen; second, with the higher temperature in the barn less air movement is required to remove a given amount of moisture from the barn. Thus either more adequate ventilation can be accomplished or smaller ventilation systems may be used to provide the necessary amount of ventilation. The use of a heat exchanger may also reduce the amount of insulation required in the barn. To determine whether this would have any value would require a complete economic analysis of the costs and benefits of the heat exchanger versus additional insulation.

In considering the use of a heat exchanger in

ventilation the following values were used in the calculations, all given as the value per cow.

Heat available	2800 Btu per hour
Moisture to be removed	3060 grains per hour
Square feet barn surface	130
Surface area in heat exchanger	10 square feet

An example of what can be accomplished by using the heat exchanger is illustrated by the following; assume that the outside temperature is 0° F. and relative humidity 50 per cent, the air then contains 2.75 grains of water vapor per pound. Calculations show that approximately 26 cfm per cow are required to maintain 75 per cent relative humidity at 40° F. inside barn temperature. As 40° F. is only an estimate as to the inside temperature that can be maintained, 30 cfm air per cow will be used in making the calculations.

With 30 cfm air flow per cow and a heat exchanger having 10 square feet of exchange surface per cow, and dimensions such that with the 30 cfm air flow the Reynolds number will be 9300, then from equation (14) U will be equal to 1.56 Btu per hour per square foot per degree Fahrenheit.

With these conditions in the Kansas State dairy barn it will be possible to maintain 35.5° F. inside temperature and 74 per cent relative humidity. Without the use of the heat exchanger and 30 cfm air flow per cow, the inside temperature will be 31.2° F. and the relative humidity 89 per cent. Condensation would occur on the walls either with or without

the heat exchanger when the wall surface temperature is 28.5° F. as the absolute humidity would be 22.5 grains of water vapor per pound of air. Calculations show that when the inside temperature is 35.5° F. and the outside temperature is 0° F. the inside surface of the walls would be 29° F., and when the inside temperature is 31.2° F. and the outside temperature is 0° F. the inside surface of the walls will be 25.4° F. Even though the 4.3° F. higher temperature possible under these conditions as shown by using a heat exchanger may seem insignificant, it does raise the temperature above the freezing point for these conditions and decreases the possibility of condensation on the walls. Fig. 3 through Fig. 6 shows the temperature difference which can be maintained in barns having various U values when there are different quantities of air flowing over a heat exchanger having 10 square feet of surface area and approximately 0.16 square feet cross sectional area per cow, and also with the same amount of air flow and no heat exchanger.

Under the above conditions with the heat exchanger there was removed from the exhaust air and added to the fresh air 23,370 Btu per hour for the 70 cows in the barn or 334 Btu per cow per hour. This is 12 per cent of the total heat given off by the cow and 39 per cent of the heat given off by the cows and not lost through the walls, windows and ceiling of the barn. In other words 39 per cent of the heat which is available for ventilation, under the assumed

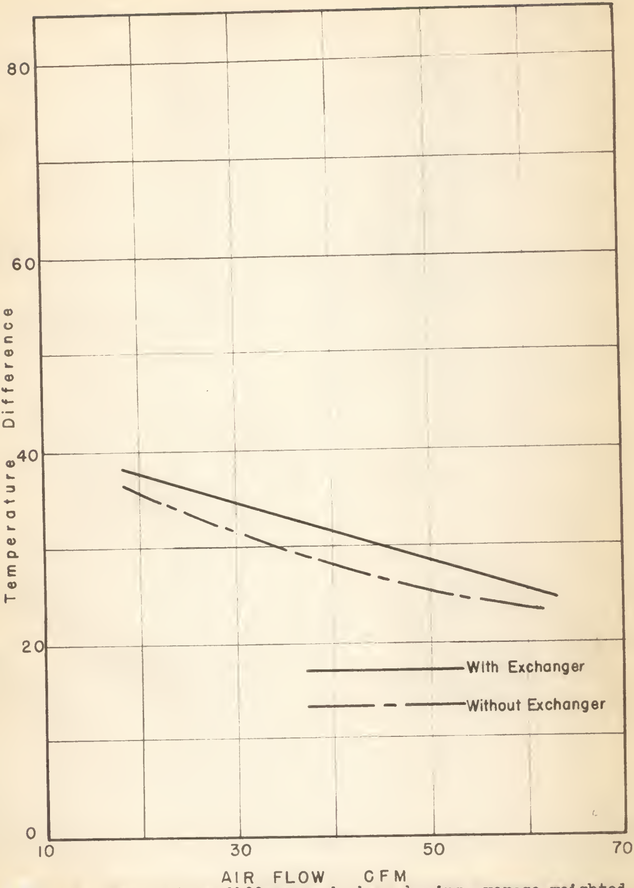


Fig. 3. Temperature difference in barn having average weighted U value of 0.43

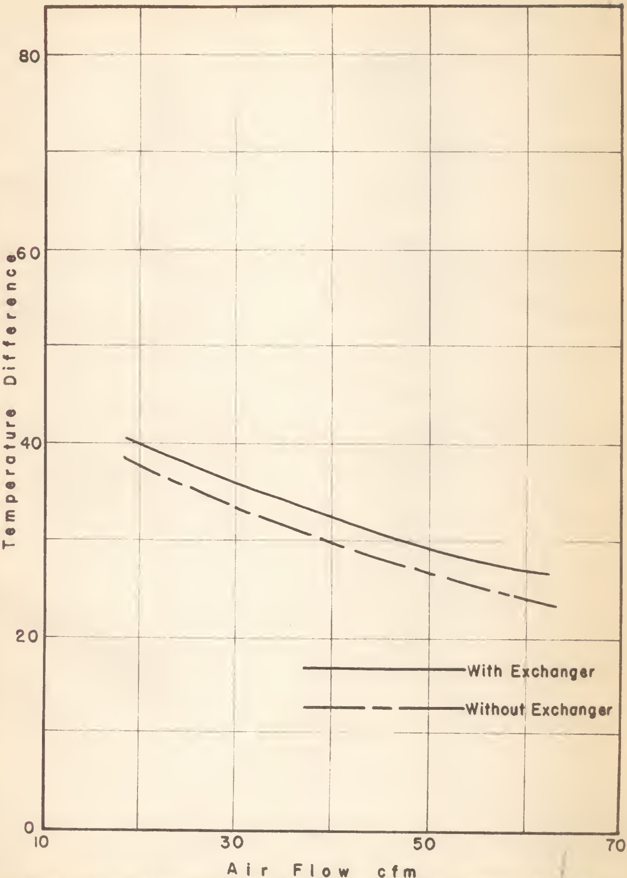


Fig. 4. Temperature difference in barn having average weighted U value of 0.40

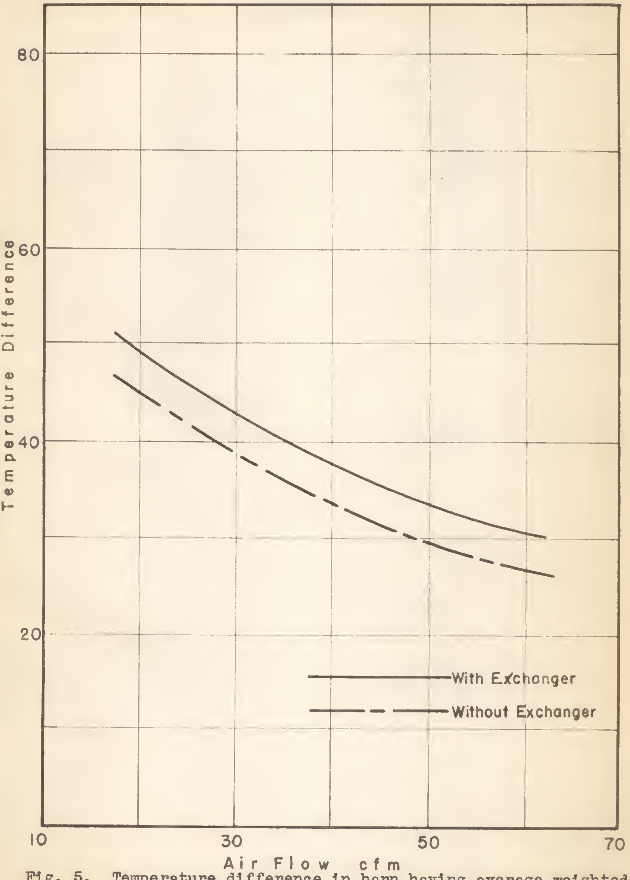


Fig. 5. Temperature difference in barn having average weighted U value of 0.30

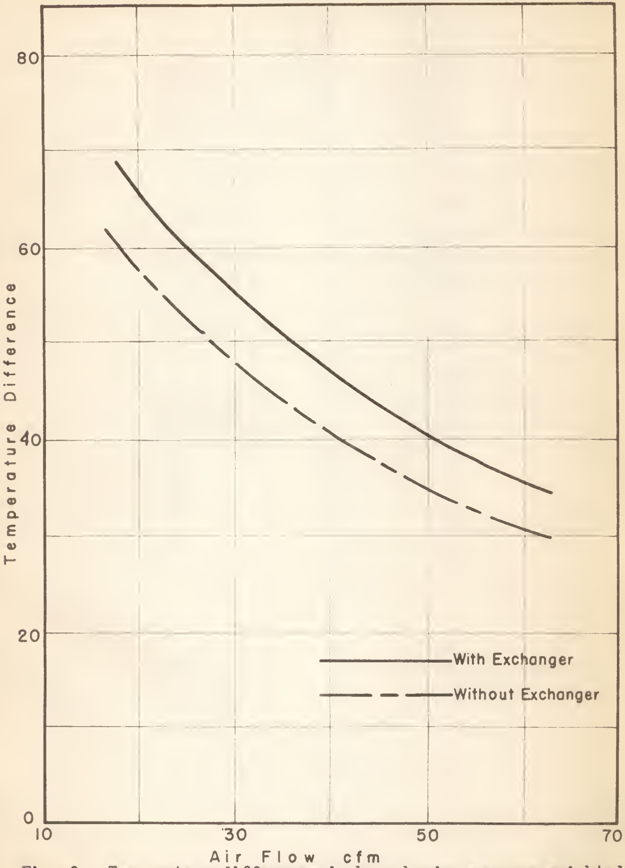


Fig. 6. Temperature difference in barn having average weighted U value of 0.20

conditions, is saved by the heat exchanger.

It has been assumed in the above calculations on the heat exchanger that all of the exhaust air as well as all of the fresh air passed through the exchanger, for it is necessary to have this occur for the exchanger to be effective. To accomplish this two fans are necessary, one to force the exhaust air and one to force the fresh air. This increases the cost of installation as well as doubling the power costs above the costs in a ventilation system having only an exhaust fan to accomplish ventilation. This means that if a heat exchanger is used the cost of installation of the ventilation system will be twice the cost of one not using a heat exchanger, plus the cost of the exchanger itself. An exchanger of the type used in the tests could be easily built by any dairyman, and the approximate cost for materials between \$3.00 and \$4.00 per cow at present prices, that is if the size is 10 square feet of exchange area per cow. So the total cost of a ventilation system using a heat exchanger would be just a little more than twice the cost of a comparable ventilating system not using a heat exchanger.

POSSIBILITIES OF USING MOW AIR FOR VENTILATION

From the data shown in Table 4, it appears that if the air for ventilation were taken from the hay mow and circulated through the barn, the barn temperatures might be raised by as much as 6 to 10 degrees Fahrenheit.

Table 4. Hay mow temperature readings and corresponding outside temperatures.

Outside temperature, °F.	Average hay mow temp. °F.	Difference
4	15	11
8	19	11
13	25	12
4	14	10
10	28	18
8	17	9
Average		11.6

However, using the mow air for ventilation would bring outside air into the mow reducing the temperatures. As the temperature is reduced, there would be more heat lost from the cow barn to the mow because of the greater temperature difference, but much of the heat could be saved by circulating the mow air through the cow barn.

The use of the mow air was not considered at the beginning of this investigation, thus sufficient data were not obtained to calculate the exact temperature difference between that and the cow barn with a given amount of ventilation using the mow air. The above data do indicate however that in the Kansas State dairy barn the use of mow air for ventilation may be very desirable. Barns where the floor

is insulated with a covering of hay or other material would not have as much advantage in raising the barn temperatures as one which is not insulated. The use of mow air and a heat exchanger should raise the cow barn temperatures considerably.

SUMMARY AND CONCLUSIONS

1. Ventilation is important in animal shelters not only because of the health and comfort of the animals and workmen, but also because of the deterioration of the structure by having moisture present. The removal of objectional odors is also an important consideration in ventilation.

2. Gravity ventilation is not being considered because of the superiority of mechanical ventilation.

3. The Kansas State dairy barn is badly in need of an improved system of ventilation, because of the excessive moisture condensation on the walls and ceiling.

4. Heat production by animals is not sufficient to provide adequate ventilation unless a large amount of insulation is provided in the structure.

5. A heat exchanger offers a means of saving some of the heat to provide for additional air change or to increase the temperature inside the structure.

6. The coefficient of heat transfer in any given heat exchanger is a function of the velocity of the air flow. With increasing velocity the turbulence of the air is increased, reducing the thickness of the air film next to

the exchange surface.

7. In the exchanger studied the coefficient of heat transfer is proportional to the 0.85 power of Reynolds number and inversely proportional to the effective diameter of the ducts in the heat exchanger.

8. Additional heat transfer surface increases the total amount of heat transfer, but to increase the surface area either the size of the exchanger must be increased or increase the number of ducts by reducing their size which increases frictional resistance to air flow.

9. In the design of a heat exchanger a balance between the effective diameter of the exchanger ducts and the surface area should be accomplished to give optimum total heat transfer with a minimum amount of frictional resistance.

10. To use a heat exchanger the structure must be relatively tight. Air leakage prevents the maximum utilization of the heat exchanger, causes a temperature drop and makes it difficult to control the system.

11. Condensation inside the structure could be reduced or eliminated by the use of a heat exchanger where, under similar conditions, this could not be accomplished without the use of an exchanger.

12. Condensation inside the exchanger will increase the coefficient of heat transfer. The effect of this was not studied and consequently was not used in heat exchanger calculations.

13. A means must be provided to drain the condensate from the exchanger and also a means for cleaning the exchanger surfaces.

14. Using a heat exchanger requires two fans instead of one, which makes twice the fan costs and twice the power requirements. In addition to this there will also be the additional cost of the exchanger itself.

15. The exchanger becomes more effective as more heat becomes available for ventilation, when the exchanger is most needed it is less effective.

16. Use of mow air for ventilation offers desirable possibilities as the average of the temperature readings in the mow were 11.6° F. above the corresponding outside temperature.

ACKNOWLEDGMENTS

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